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NASTRAN APPLICATION FOR THE PREDICTION OF AIRCRAFT INTERIOR NOISE

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Abstract

The application of a structural-acoustic analogy within the NASTRAN finite element program for the prediction of aircraft interior noise is presented. The reliability of the procedure is assessed through an analysis of the principle components involved in the fluid-structure coupling, and is compared to simple structures with known theoretical results. Some refinements of the method, which reduce the amount of computation required for large, complex structures, are then discussed. Finally, further improvements are proposed and preliminary comparisons with structural and acoustic modal data obtained for a large composite cylinder, are presented.

Introduction

The prediction and reduction of aircraft interior noise are important considerations for conventional propeller aircraft now entering the commercial market as well as for aircraft currently being developed, such as the advanced turboprop. Consequently, the interior noise problem is receiving attention even during the first stages of the aircraft design process.[1,2] Also, the need for laboratory tests on full scale models to validate new theoretical prediction methods has been recognized.[3,4] The theoretical approach has progressed through several stages, beginning with very simple models of the aircraft fuselage[5,6] and proceeding to very detailed methods and computer programs which discretize the structure and the interior acoustic volume and define the coupling characteristics therein. Among the several analytical methods available, the finite element method has been chosen for this study for several reasons. It is fully documented, available worldwide, and can be used to model complex structural and acoustic geometries. The theory which defines the finite element solution for fluid-structure interaction problems is available in the literature,[7-12] as is the practical application using the NASTRAN code.[13-17] Only recently has this approach been applied to aircraft structures.[18,19] An analysis of the fluid-structure interaction problem based on the finite element method and NASTRAN is presented in this paper and compared with studies in the literature. The initial results are very promising, however, some refinement of the numerical techniques may be necessary (especially for large structures with many degrees of freedom) in order to reduce computational costs and provide a cost-effective tool for use during the final definition phase of aircraft design. This paper also presents preliminary numerical predictions using both the structural and acoustic finite element models to describe an actual aircraft fuselage model available in the laboratory of the Acoustics Division of the NASA Langley Research Center. These predictions are in good agreement with experimentally obtained results. Finally, refinements of the NASTRAN model and plans for future work are discussed.

Symbols

$[A]$	area matrix for fluid-structure coupling (equation 13)
$[B]$	material definition matrix (equation 5)
c	speed of sound
E	Young's modulus
$e^{i\omega t}$	time dependence of the harmonic forcing function
F_o	amplitude of the harmonic forcing function
$\{F_s\}$	external forcing vector
G	tangential elasticity modulus
i	imaginary unit $= -1$
$[K]$	stiffness matrix
$[M]$	mass matrix
n	normal vector (positive outward)
p	pressure variation from equilibrium pressure value
t	time variable
u_x	displacement in the x-direction
∇^2	Laplacian operator. In cartesian coordinates $\frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} + \frac{\partial^2}{\partial z^2}$
$\frac{\partial}{\partial x}, \frac{\partial}{\partial y}, \frac{\partial}{\partial z}, \frac{\partial}{\partial n}$	partial derivative with respect to x, y, z, or n
$\frac{\partial^2}{\partial x^2}, \frac{\partial^2}{\partial y^2}, \frac{\partial^2}{\partial z^2}$	second partial derivative with respect to x^2, y^2, or z^2
$\{\varepsilon\}$	strain vector
λ, μ	Lamé's constants
ν	Poisson's ratio

ρ	mass density
$\{\sigma\}$	stress vector
σ_{xx}	axial stress in the x-direction
τ_{xy}, τ_{yz}, τ_{xz}	shear stress in the x-y, y-z, or x-z plane
ω	angular frequency of harmonic forcing function
$[\ ]^T$	transpose matrix
$\ddot{}$	double dot. Second time derivative

Subscripts

a	acoustic
n	normal direction
s	structure

Structural-Acoustic Analogy

It is possible to solve acoustic problems using structural code which already exists in the Finite Element Method. The technique is based on a structural-acoustic analogy which relates structural displacement to acoustic pressure. Specific problems have been solved using this approach,[7-19] and the theoretical development has been well documented. In this paper the fundamental steps are included for the sake of clarity.

Theory

The scalar acoustic wave equation in terms of the variation of pressure from the equilibrium pressure is

$$\nabla^2 p = \frac{1}{c^2} \frac{\partial^2 p}{\partial t^2} \tag{1}$$

which, in Cartesian coordinates is

$$\frac{\partial^2 p}{\partial x^2} + \frac{\partial^2 p}{\partial y^2} + \frac{\partial^2 p}{\partial z^2} = \frac{1}{c^2} \frac{\partial^2 p}{\partial t^2} \tag{2}$$

The equation governing the equilibrium of stresses in a material in a particular direction (x, for example) is

$$\frac{\partial \sigma_{xx}}{\partial x} + \frac{\partial \tau_{xy}}{\partial y} + \frac{\partial \tau_{xz}}{\partial z} = \rho_s \frac{\partial^2 u_x}{\partial t^2} \tag{3}$$

Equations (2) and (3) are mathematically similar, and an "analogy" can be obtained if

$$\sigma_{xx} = \frac{\partial p}{\partial x} ; \; \tau_{xy} = \frac{\partial p}{\partial y} ; \; \tau_{xz} = \frac{\partial p}{\partial z} \quad \rho_s = \frac{1}{c^2} ; \; u_x = p \tag{4}$$

Thus, it is possible to solve acoustic problems using existing structural analysis codes based on the displacement formulation of the Finite Element Method, in particular NASTRAN.[20,21]

In order to complete the analogy and give practical ideas as to its NASTRAN application, consider the general stress-strain relationship,

$$\{\sigma\} = [B] \; \{\varepsilon\} \tag{5}$$

If we consider the displacement in the x-direction only ($u_y = u_z = 0$), equation (5) can be written in terms of the relations (4) as

$$\begin{Bmatrix} \sigma_{xx} \\ \sigma_{yy} \\ \sigma_{zz} \\ \tau_{xy} \\ \tau_{yz} \\ \tau_{xz} \end{Bmatrix} = \begin{bmatrix} 1 & B_{12} & B_{13} & 0 & B_{15} & 0 \\ & B_{22} & B_{23} & 0 & B_{25} & 0 \\ & & B_{33} & 0 & B_{35} & 0 \\ & & & 1 & B_{45} & 0 \\ & \text{SYMM.} & & & B_{55} & 0 \\ & & & & & 1 \end{bmatrix} \begin{Bmatrix} \partial u_x/\partial x \\ 0 \\ 0 \\ \partial u_x/\partial y \\ 0 \\ \partial u_x/\partial z \end{Bmatrix} \tag{6}$$

where the terms B_{ij} are arbitrary. However, it is often convenient to choose B_{ij} so that the matrix [B] is isotropic and therefore invariant for any coordinate system. This can be achieved in NASTRAN by using the card which defines a linear, temperature-independent, isotropic material (MAT1 card), and substituting the proper values for the material constants. For example, the material definition matrix, [B], for the three-dimensional problem[20] is

$$[B] = \begin{bmatrix} \lambda+2\mu & \lambda & \lambda & 0 & 0 & 0 \\ & \lambda+2\mu & \lambda & 0 & 0 & 0 \\ & & \lambda+2\mu & 0 & 0 & 0 \\ & & & \mu & 0 & 0 \\ & \text{SYMM} & & & \mu & 0 \\ & & & & & \mu \end{bmatrix} \tag{7}$$

where $\lambda + 2\mu = 1$ and $\mu = 1$ in order to obtain the general matrix given by equation (6). Since the Lamé constants λ and μ are

$$\lambda = \frac{E\nu}{(1+\nu)(1-2\nu)} \quad ; \quad \mu = G = \frac{E}{2(1+\nu)} \tag{8}$$

the following values input with the MAT1 card

$$E = 1 \times 10^{20} \; ; \; G = 1 \; ; \; \nu = .5 \times 10^{20} \; ; \; \rho = \frac{1}{c^2} \tag{9}$$

define the acoustic analogy. Similarly, the [B] matrix for the two-dimensional case is

$$[B] = \frac{E}{1-\nu^2} \begin{bmatrix} 1 & \nu & 0 \\ \nu & 1 & 0 \\ 0 & 0 & \frac{(1-\nu)}{2} \end{bmatrix} \tag{10}$$

and the corresponding material properties are input as

$$E = 2 \times 10^{-6} \; ; \; G = 1 \; ; \; \nu = -.999999 \; ; \; \rho = \frac{1}{c^2} \tag{11}$$

Using this approach, equation (1) can be solved for many cases, and proves especially useful when the geometry of the enclosure is irregular and cannot be studied adequately using known results for simple geometries.

Finite Element Formulation and Validation Examples

The equation of motion of an acoustic enclosure with rigid walls and no forcing function can be written in terms of the Finite Element notation as

$$[M_a]\{\ddot{p}\} + [K_a]\{p\} = \{0\} \tag{12}$$

where $[M_a]$ is the acoustic 'mass' matrix, $[K_a]$ is the acoustic 'stiffness' matrix and $\{p\}$ is the vector of pressure values at the grid points. If the pressure is harmonic, equation (12) becomes a classical eigenvalue problem that can be solved using standard NASTRAN methods and the acoustic resonance frequencies and acoustic mode shapes for any geometry can easily be extracted. The validity of this formulation has been studied for cavities with simple geometries which have straightforward theoretical solutions. For the present paper two particular geometries have been studied, one having a rectangular cross section and one with a circular cross section. A comparison of the acoustic mode shapes and resonance frequencies for the 2-dimensional cross section and the 3-dimensional volume derived theoretically and predicted using the present NASTRAN formulation is tabulated in Table I. These data are in general good agreement, with some small differences at higher modes where a finer mesh may be necessary. For the NASTRAN formulation the QUAD2 membrane element was used to model the cross-sectional area for the 2-dimensional case and the HEXA2 solid element was used to model the 3-dimensional acoustic volume. Figure 1 shows some of the acoustic mode shapes predicted using the NASTRAN formulation and plotted using PATRAN-G post-processing capabilities.[22]

Fluid-Structure Interaction

Theory

A complete description of the fluid-structure interaction problem, in terms of finite element models of the structure and the enclosed acoustic volume, is given by the following coupled equation of motion

$$\begin{bmatrix} M_s & 0 \\ -(\rho c)^2 A^T & M_a \end{bmatrix} \begin{Bmatrix} \ddot{u} \\ \ddot{p} \end{Bmatrix} + \begin{bmatrix} K_s & A \\ 0 & K_a \end{bmatrix} \begin{Bmatrix} u \\ p \end{Bmatrix} = \begin{Bmatrix} F_s \\ 0 \end{Bmatrix} \tag{13}$$

where the matrix, [A], ensures the proper coupling between structural and acoustic models. At a fluid-structure interface the boundary condition is

$$\frac{\partial p}{\partial n} = -\rho \ddot{u}_n \tag{14}$$

where n is the normal, positive outward, unit vector at the interface and ρ is the mass density of the fluid. For the present finite element application, the force exerted by the structure on the fluid is $(\rho c)^2 A\ddot{u}_n$ at each grid point located along the fluid-structure interface.[14] Here, A is the surface area associated with the grid point and u_n is the normal component of the fluid particle acceleration. The force of the fluid acting on the structure is expressed as a surface pressure force equal to -pA applied to each interface structural grid point.

The general equation of motion, therefore, can be written as

$$[M_s]\{\ddot{u}\} + [K_s]\{u\} = \{F_s\} - [A]\{p\} \tag{15}$$

for the structural model and

$$[M_a]\{\ddot{p}\} + [K_a]\{p\} = (\rho c)^2 [A]^T \{\ddot{u}\} \tag{16}$$

for the acoustic model. The matrix [A] must be entered by the user since it is not included in the standard, rigid format of COSMIC/NASTRAN. It is a sparse (n x m) matrix whose non-zero elements correspond to the fluid-structure interface locations. The non-zero elements are the lumped areas at the designated interface grid points and can be extracted from the OLOAD output which results from the application of a unit pressure level to the structure surface.[12] These values are entered in the appropriate locations of the complete mass and stiffness matrices using the DMIG (Direct Matrix Input at Grid points) card available in NASTRAN.

For the current procedure, only the coupling terms for the stiffness matrix need to be input using the DMIG cards. The coupling terms for the mass matrix are computed by the simple alteration sequence (ALTER) shown in figure 2, which transposes the [A] matrix and multiplies it by $-(\rho c)^2$.

The solution to the fluid-structure interaction problem can be obtained using a direct or a modal approach. The direct approach solves equation (13) directly with NASTRAN[20,21] for the frequencies or grid points selected by the user. This solution can be retained for comparison with other more approximate solutions which modify the basic equations in order to reduce the computation time.

The modal approach to the fluid-structure interaction problem can also be used within the NASTRAN framework.[11] First, the eigenvalues and eigenvectors of equation 13 are computed while ignoring the coupling terms. Then the modal content of the coupling matrices is extracted and combined with the previously obtained modal mass and stiffness matrices. These new matrices then describe the entire problem. In order to reduce the computation time required to solve these matrix equations it is necessary to reduce the size of the matrices. This can be accomplished by considering only the significant modes in a specified frequency range and disregarding those modes which do not appreciably influence the solution. However, these modes must be selected carefully to avoid unacceptable errors, such as those shown in reference 19.

Another approach which reduces the computational effort,[19] untested in the present study, is based on the assumption that the effects of the acoustic medium on the structure are negligible. If this is true equation 13 can be split into equations 15 and 16 which can then be decoupled, since the pressure forcing term, $[A]\{p\}$, on the right hand side of equation 15 is negligible with respect to the structural forcing terms, $\{F_s\}$. In this way the accelerations can be computed from equation 15 and input to equation 16 to obtain the corresponding pressure values. These uncoupled equations can be solved with NASTRAN based on a direct, modal, or mixed approach, depending on the computational evidence.

Examples

In order to more completely understand the fluid-structure interaction problem, as defined according to the NASTRAN formulation above, some simple models have been developed and compared with studies existing in the literature.[18,19]

The basic acoustic model is the same as described previously -- a cylinder with a circular cross section. The structural model was designed to fit with the acoustic model. The cylinder's dimensions are 914.4 mm (radius) by 25.4 mm (length) with a skin thickness of 0.8128 mm. These dimensions are needed to compute the BAR element used to model a structural strip. A point force of the form $F=F_0e^{i\omega t}$ (F_0=.4534 kgf) was applied to the structure. The frequency response resulting from this forcing function is shown in Figures 3 and 4. Figure 3 shows the frequency response of the structure at the point where the force is applied, and compares predictions using both the direct and modal approach. Figure 4 shows the frequency response at three points in the acoustic volume, A, B, and C: the interface point where the force is applied, a point at one-half the radius, and a point on the cylinder axis (zero radius). Again, predictions based on the direct and modal approach are compared. In order to depict the three-dimensional response, the post-processing program PATRAN-G, available at NASA Langley Research Center, has been used. Figure 5 shows some of the cylinder's responses for certain specific frequencies. These results are in good agreement with those found in the literature[18] with some minor discrepancies due to differences in the corresponding mesh size and damping characteristics.

The three-dimensional fluid-structure interaction problem has not been completed because the amount of computation time was considered too large and costly for the immediate needs of this preliminary study. Table II lists the number of degrees of freedom and corresponding CPU time (using the CDC CYBER 855 computer at NASA LaRC) for the models in the present study. Obviously, the large increase in CPU time required to go from the basic acoustic problem to the fluid-structure interaction problem is not only caused by the increased complexity of the solution but is also related to the increase in number of degrees of freedom required to describe the problem.

Experimental Application

In order to further validate the finite element model presented in the previous paragraphs, a real structure was modeled and predictions were compared to measured data. The test structure is a cylindrical fuselage model under study in the laboratory of the Acoustics Division at NASA Langley Research Center. It is a stiffened, filament wound composite cylinder 838.2 mm in diameter and 3657.6 mm in length with a skin thickness of 1.7 mm. Additional details of the cylinder's geometric properties are available in reference 4. Figure 6 shows some of the mode shapes predicted from the structural and acoustic finite element models for the two- and three-dimensional cases. An experimental modal analysis of the test cylinder's structure and enclosed volume has been performed.[4] Table III lists the mode shapes and frequencies predicted by the finite element model and measured on the test cylinder. These results indicate that the lowest structural modes are dominated by the frame vibration and the individual panels behave like simple lumped masses. Thus the two-dimensional structural model agrees quite well with the experimental results and no significant improvements are obtained from the three-dimensional, one bay model. The acoustic behavior is different. The first few acoustic modes are dominated by the largest dimension, the cylinder length, so the three-dimensional finite element model yields better agreement with the experimental results than the two dimensional model.

Also included in Table III are predictions of the structural and acoustic modal frequencies obtained from the Propeller Aircraft Interior Noise (PAIN) model.[23] The PAIN program was developed to predict aircraft interior noise based on assumed functions for the structural modes and a finite difference formulation for the modes of the enclosed acoustic volume. Comparisons of data from the different prediction models indicate that the two-dimensional model may be sufficient for studying trends but the three-dimensional finite element acoustic model is required for practical applications. As stated previously, the computer time must be reduced to allow cost effective studies of different geometric configurations. One possibility would be to couple the two-dimensional structural model to the three dimensional acoustic model, especially in the region of low structural modal density. Another technique to reduce the computer time is to use the uncoupled solution presented in reference 19, that has been briefly described above.

Concluding Remarks

A NASTRAN finite element application has been presented which can predict the interior noise of an aircraft fuselage. The principal theoretical steps have been presented and comparisons between the numerical predictions and exact theoretical results for simple structures have shown the method's practicality, especially in the low modal density region. The coupled fluid-structure interaction problem, implemented with NASTRAN, was then described and preliminary results show good agreement with the available literature. Finally, validation of the finite element structural and acoustic models has been obtained through comparison with experimental data obtained on a laboratory cylinder. Ongoing analysis of the finite element model will indicate the feasibility of the uncoupled solution for large structures, and will study the effects of various parameters on the interior noise level, such as cabin pressurization, damping, structural modifications,[24] etc. These are important aspects which can be studied easily due to the great flexibility of the finite element model.

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TABLE I. ACOUSTIC NATURAL FREQUENCIES FOR TWO-AND THREE-DIMENSIONAL ENCLOSURES. COMPARISON BETWEEN THEORY AND NASTRAN RESULTS.

2-D					
Square Enclosure (1000 mm x 1000 mm)			Circular Enclosure (R = 914.4 mm)		
	Frequency (Hz)			Frequency (Hz)	
Mode	Theory	NASTRAN	Mode	Theory (Ref. 19)	NASTRAN
0,0	0.00	0.00	0,0	0.00	0.00
0,1	170.10	169.40	1,0	109.05	108.39
1,0	170.10	169.40	2,0	180.83	178.64
1,1	240.56	239.57	0,1	226.86	224.77
2,0	340.21	334.64	3,0	248.74	237.19
0,2	340.21	334.64	4,0	314.82	289.52
2,1	380.36	375.07	1,1	315.64	313.11
1,2	380.36	375.07			
2,2	481.12	473.25			

3-D					
Rectangular Enclosure (1000 mm x 1000 mm x 10000 mm)			Cylindrical Enclosure (R = 914.4 mm, L = 4064 mm)		
	Frequency (Hz)			Frequency (Hz)	
Mode	Theory	NASTRAN	Mode	Theory (Ref. 19)	NASTRAN
0,0,0	0.00	0.00	1,0,0	41.85	41.77
0,0,1	17.01	16.92	1,1,0	116.77	116.20
0,0,2	31.02	33.33	3,0,0	125.55	123.20
0,0,3	51.03	48.73	3,1,0	166.27	164.90
0,0,4	68.04	62.65	1,2,0	185.61	183.42
.	.	.	5,0,0	209.25	198.46
.	.	.			
.	.	.			
1,0,0	170.10	165.77			
0,1,0	170.10	165.77			
1,0,1	170.95	166.58			
0,1,1	170.95	166.58			

TABLE II. NUMBER OF DEGREES OF FREEDOM (DOF) AND CPU TIME (CDC CYBER 855) FOR SOME SIMPLE AND COMPLEX NASTRAN FINITE ELEMENT MODELS.

TYPE OF COMPUTATION	NASTRAN FINITE ELEMENT MODEL	DEGREES OF FREEDOM	CPU TIME (SEC.)
Normal Mode Analysis (Acoustic part only)	2-D rectangular cross section	121	40
	3-D rectangular volume	250	120
	2-D circular cross section	89	18
	3-D cylindrical volume	546	300
Fluid-Structure Coupling (Direct Approach)	2-D circular cross section	228	360
	3-D cylinder	991	--- (unavailable)

TABLE III. **NATURAL FREQUENCIES OF COMPOSITE CYLINDERS. COMPARISON OF MEASURED DATA TO NASTRAN AND PAIN PREDICTIONS FOR THE STRUCTURAL AND ACOUSTIC MODELS.**

Structure

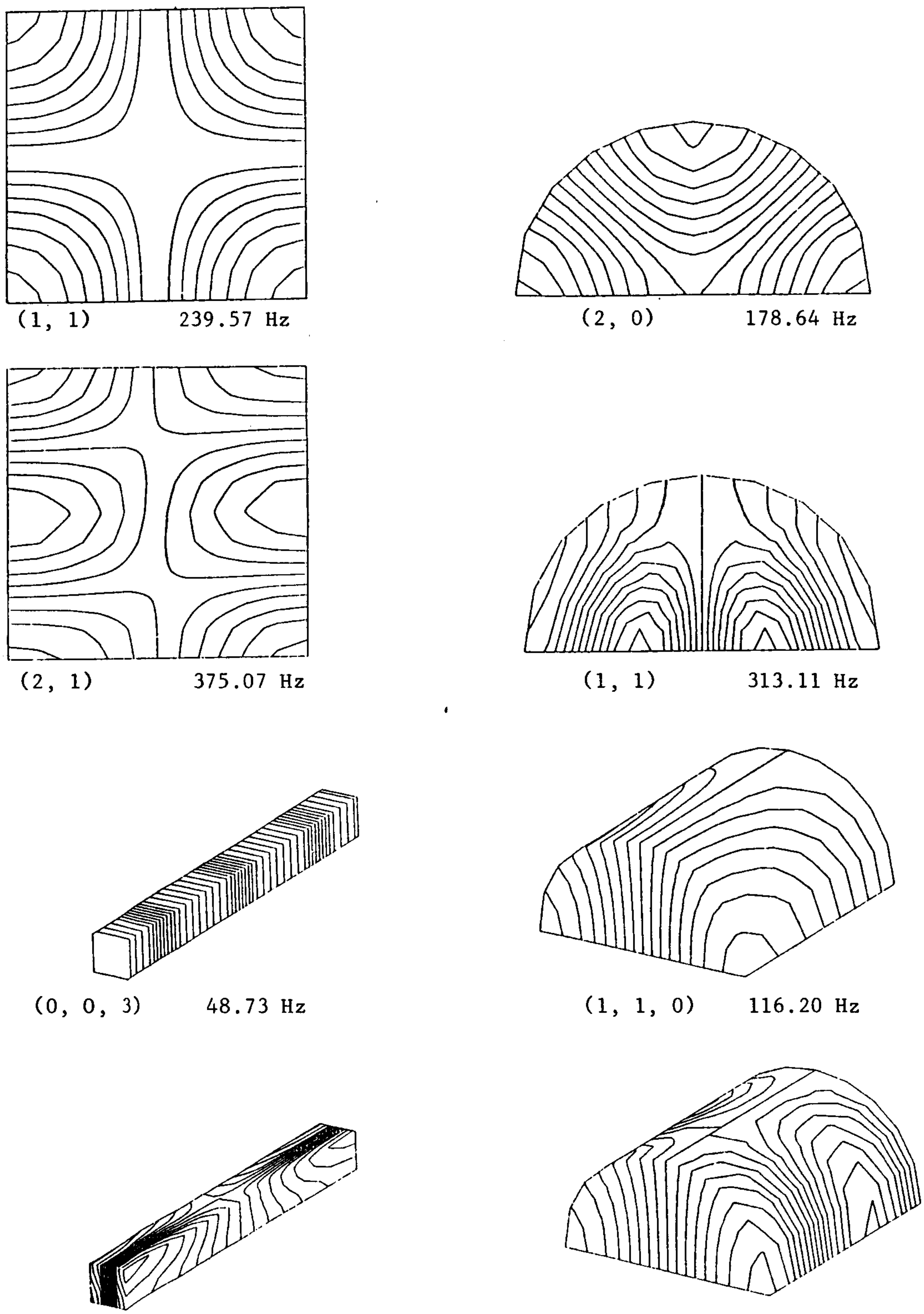
Mode Shape (1, c)		Frequency (Hz)		
		NASTRAN 2-D Model	NASTRAN 3-D Model (one bay)	Experiment
0,2	A	29.34	31.58	--
0,2	S	48.50	54.53	46
0,3	A	89.40	96.07	84
0,4	S	131.38	134.34	121
0,4	A	149.07	156.67	--
1,4	S	148.16	162.40	--
0,5	A	232.81	250.67	--

A — antisymmetric
1 — longitudinal
S — symmetric
c — circumferential

Acoustic Volume

Mode Shape (1, h, v)*	Frequency (Hz)			
	NASTRAN 2-D Model	NASTRAN 3-D Model (full length)	Experiment	PAIN Program
1,0,0	---	49	47	48
2,0,0	---	97	94	96
1,1,0	109	121	121	125
3,0,0	---	142	143	143
2,1,0	---	148	150	150
0,0,1	154	161	171	160
3,1,0	---	183	183	184
4,0,0	---	---	191	191
3,0,1	---	218	216	215
1,2,0	197	232	230	228
5,0,0	---	233	239	239

* 1 — longitudinal, h — horizontal, v — vertical



(1, 1) 239.57 Hz

(2, 0) 178.64 Hz

(2, 1) 375.07 Hz

(1, 1) 313.11 Hz

(0, 0, 3) 48.73 Hz

(1, 1, 0) 116.20 Hz

(1, 0, 1) 166.58 Hz

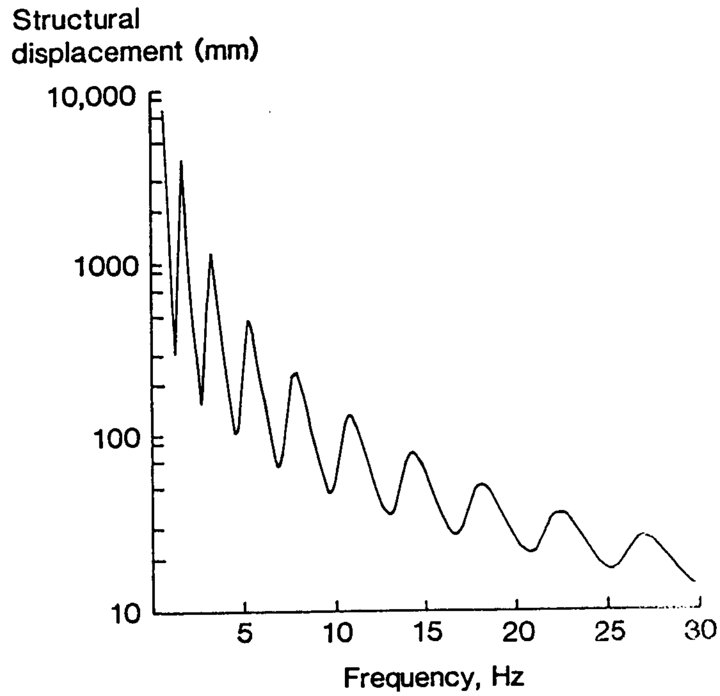
(3, 1, 0) 164.90 Hz

Figure 1. Examples of acoustic mode shapes for some simple geometries predicted with NASTRAN using a structural-acoustic analogy.

```
ID FIR,ACO
TIME 100
APP DISP
SOL 8,0
ALTER 100,100 $
MTRXIN CASEXX,MATPOOL,EQDYN,,TFPOOL/K2DPP,,B2PP/LUSETD/S,N,
       NOK2DPP/S,N,NOB2PP $
ALTER 100 $
LABEL  LBLSAV $
TRNSP  K2DPP/XMP $
ADD    XMP,/M2DPP/C,Y,ALPHA=(-1.81-15,0.0)/C,Y,BETA=(0.0,0.0) $
PARAM  //C,N,NOP/V,N,NOM2DPP=1 $
ENDALTER
CEND
```

```
ID FIR,ACO
TIME 100
APP DISP
SOL 11,0
ALTER 91,91 $
MTRXIN  CASEXX,MATPOOL,EQDYN,,TFPOOL/K2PP,,B2PP/LUSETD/S,N,
        NOK2PP/S,N,NOB2PP $
ALTER 91 $
LABEL   LBLSAV $
TRNSP   K2PP/XMP $
ADD     XMP,/M2PP/C,Y,ALPHA=(-1.81-15,0.0)/C,Y,BETA=(0.0,0.0) $
PARAM   //C,N,NOP/V,N,NOM2PP=1 $
ENDALTER
CEND
```

Figure 2. ALTER sequence for solution of fluid-structure interaction. Direct approach - COSMIC/NASTRAN solution 8. Modal approach - COSMIC/NASTRAN solution 11.



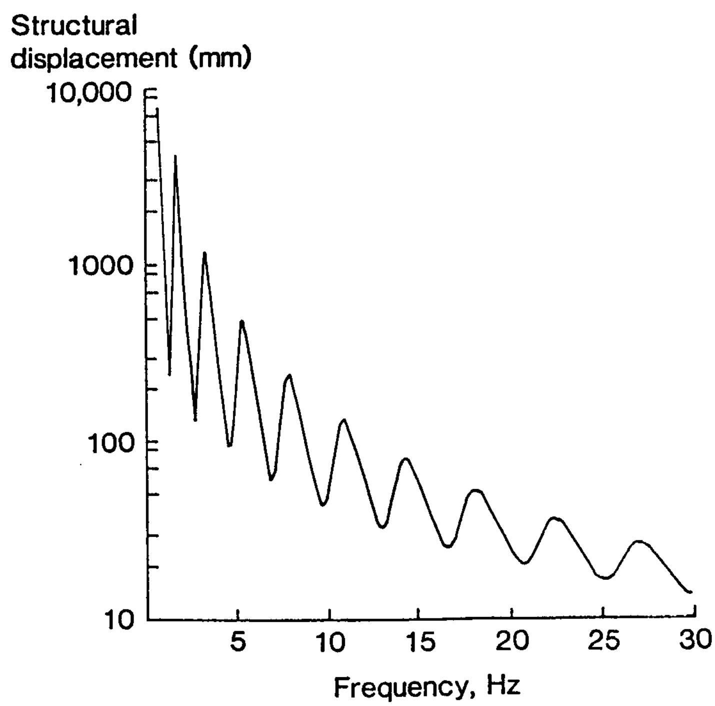


Figure 3. Frequency response function of structural point located directly under load. (a) NASTRAN direct approach, (b) NASTRAN modal approach.

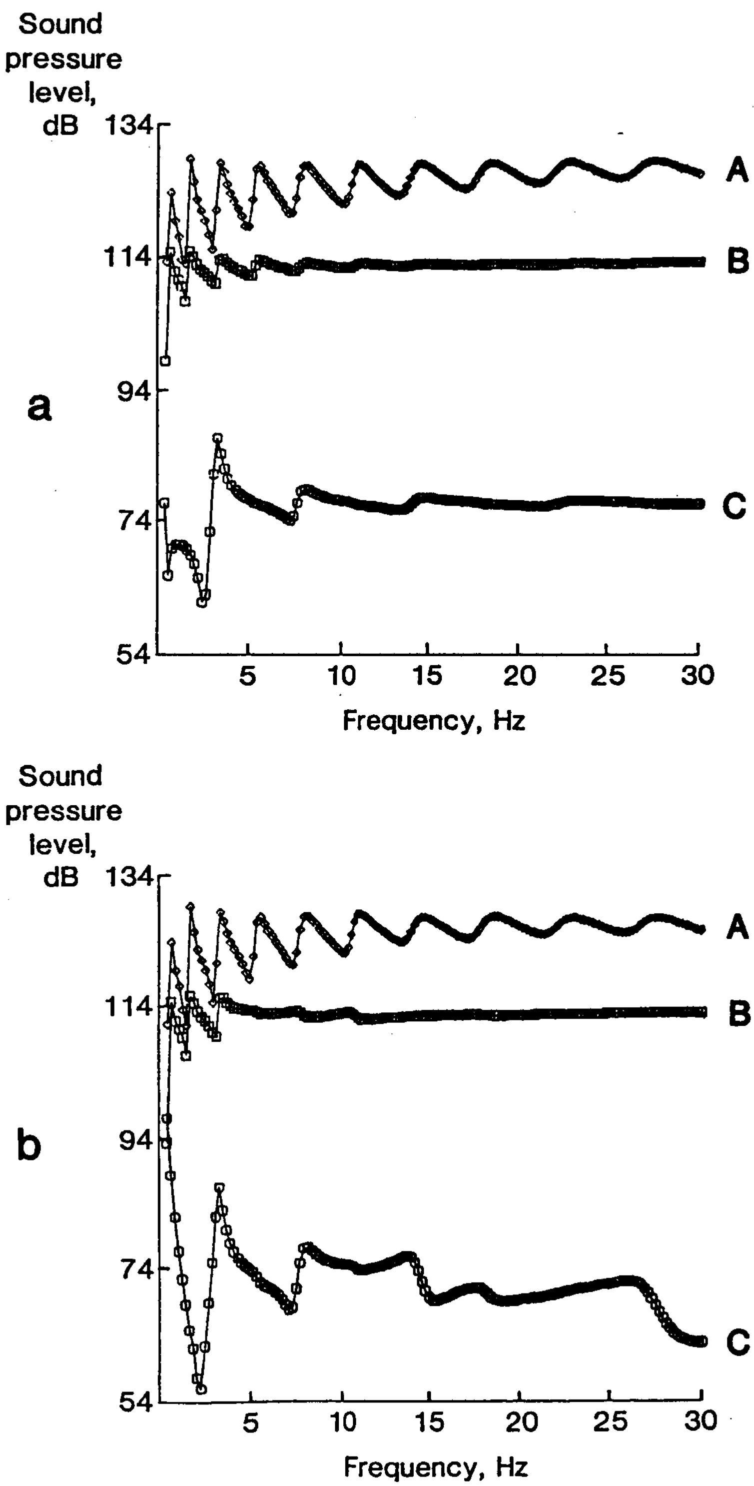


Figure 4. Acoustic frequency response: A = acoustic - structure interface, B = half-radius, C = center, (a) NASTRAN direct approach, (b) NASTRAN modal approach

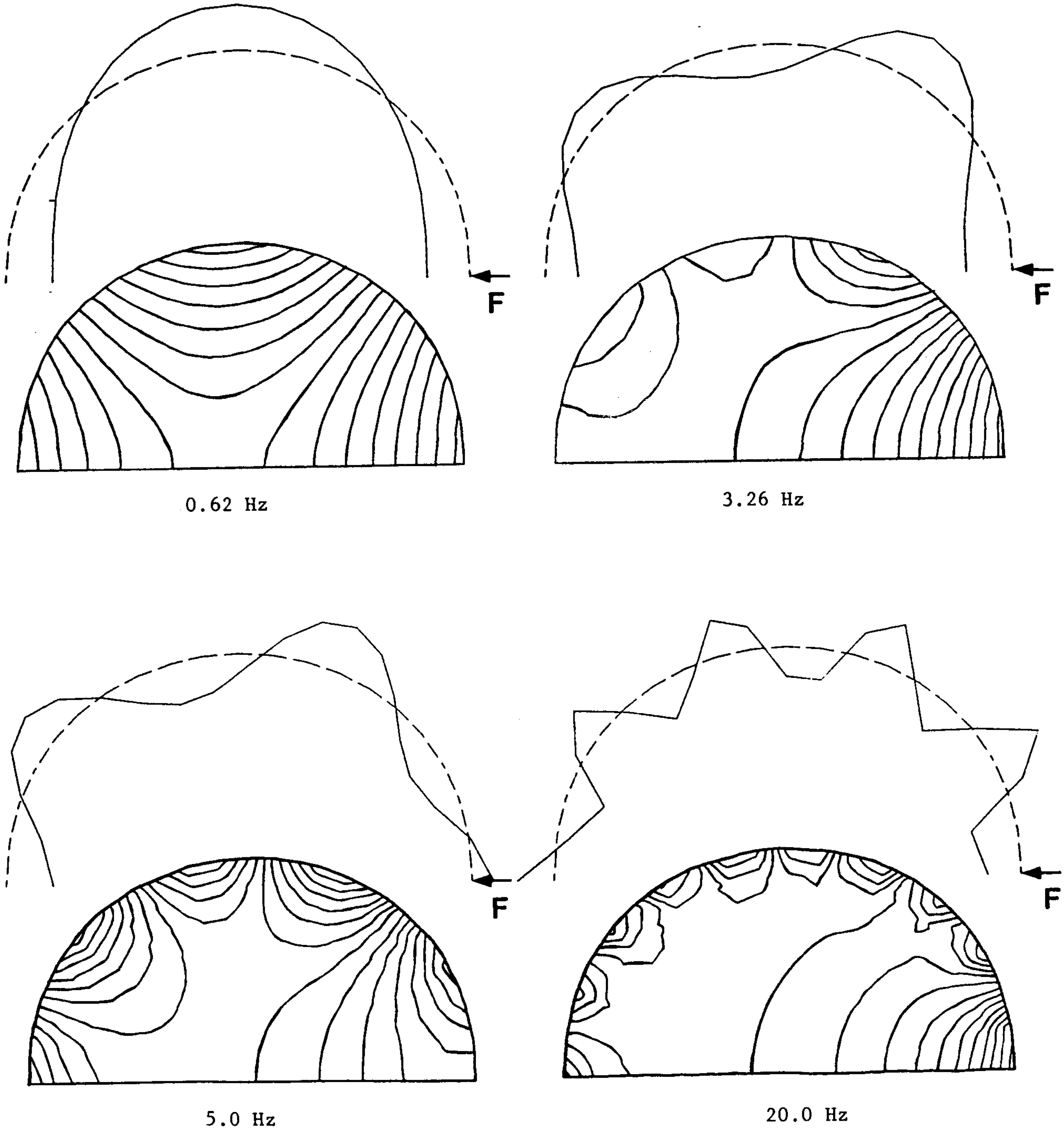


Figure 5. Dynamic shapes of structure and acoustic space predicted with NASTRAN using 2D model and fluid-structure interaction equations. Frequency indicated is for structure.

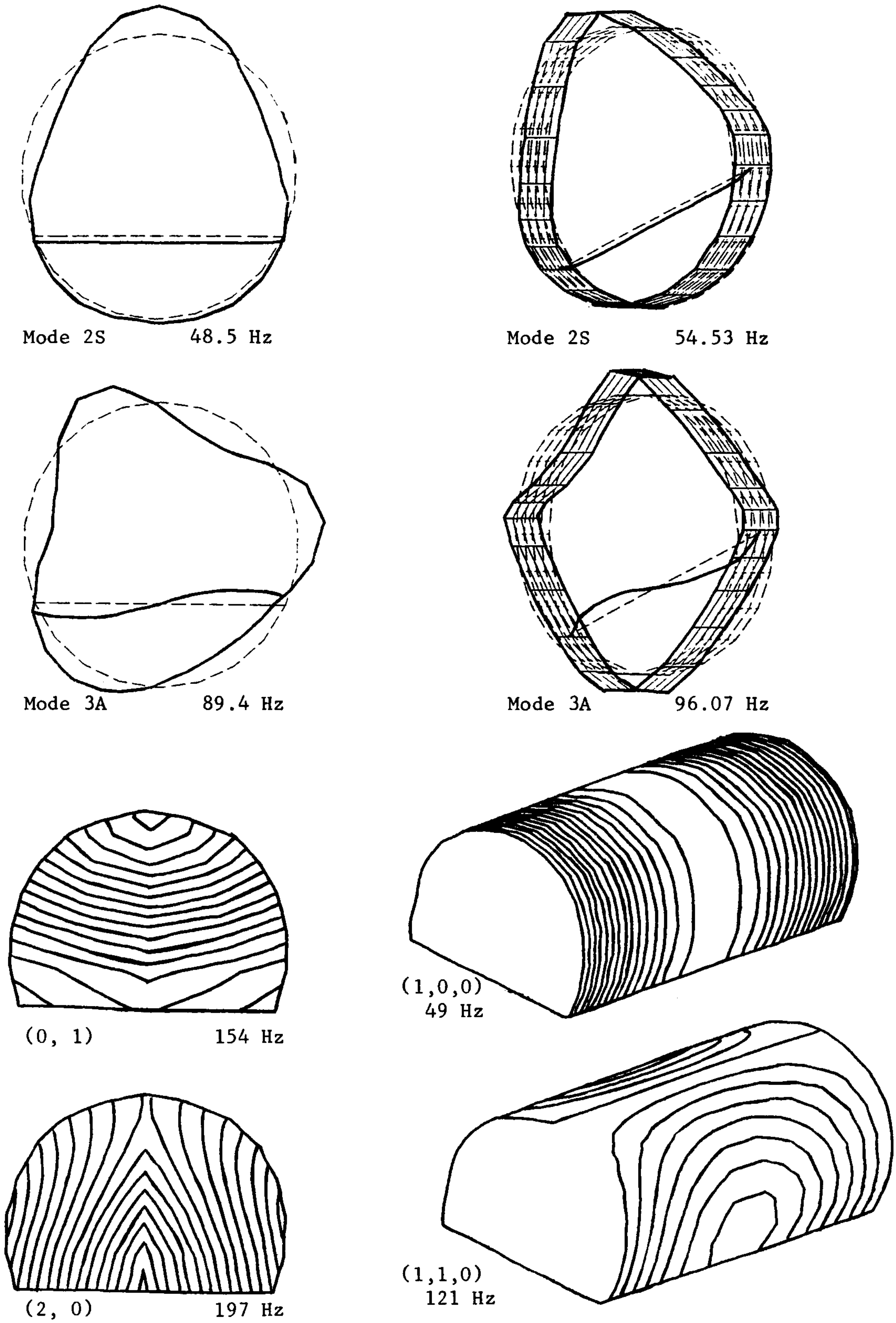


Figure 6. NASTRAN predictions of structural and acoustic mode shapes of composite cylinder based on two- and three-dimensional models.